Elsevier Editorial System(tm) for Energy and Buildings
Manuscript Draft

Manuscript Number: ENB-D-12-01052R2

Title: In situ optimization methodology for the water circulation pumps frequency of ground source heat pump systems

Article Type: Full Length Article

Keywords: Heating and cooling systems, ground source heat pump, energy efficiency

Corresponding Author: Dr. Carla Montagud,

Corresponding Author's Institution: IIE

First Author: Carla Montagud, Ph.D

Order of Authors: Carla Montagud, Ph.D; José Miguel Corberán, Professor; Álvaro Montero, Ph.D

Abstract: This paper presents an experimental study of the influence of the water circulation pumps frequency at the indoor and outdoor loops in the overall system performance of a ground source heat pump monitored plant, and describes a three steps new methodology to optimize the system energy performance and obtain the values of the optimal frequencies when ON/OFF regulation is employed. The first step consists of several experimental tests of pseudo-random sequence of frequency steps for both, internal and external circulation pumps, carried out during a single day. The second step is the characterisation of the quasi-steady state system performance as a function of circulation pumps frequencies. The final step consists of, by means of an analytical expression, the extrapolation of results to any partial load ratio, what allows finding the optimal frequency as a function of the instantaneous thermal load. The advantage of the proposed methodology is that it can be carried out on site and is able to consider the phenomena ocurring at the heat pump and the ground source heat exchanger when the flow rates are varied. The methodology can be applied to any installation incorporating variable speed circulation pumps. Results indicate large energy savings potential for this kind of installation.

Title: In situ optimization methodology for the water circulation pumps frequency of ground source heat pump systems.

Authors names in desired order and contact email:

C. Montagud[a*], carmonmo@iie.upv.es

J. M. Corberán[a], corberan@ter.upv.es

Á. Montero[a], monteroalvaro@hotmail.com

Affiliations:

[a] Instituto de Ingeniería Energética, Universidad Politécnica de Valencia, Camino de vera s/n,46022,Valencia,Spain.

[*]Corresponding author. Tel.: +34 963879910; fax: +34 963877329. E-mail address: carmonmo@iie.upv.es (C. Montagud)

Comments

We are presenting in the paper a new methodology which is based in a long research campaign and that we propose to be applied in practice, so we feel that it is necessary to present it in detail and carefully justify everything.

Finally, I take this chance to show my gratitude for you taking the time to review the manuscript. Hoping it will be successfully accepted and published, I remain at your disposal to provide any further documentation required to clarify any issue aroused.

Sincerely yours,

Carla Montagud Montalvá, Ph.D.
Instituto de Ingeniería Energética
(Institute for Energy Engineering)

UNIVERSIDAD POLITÉCNICA DE VALENCIA
Camino de Vera s/n, Edificio 8E, bloque F, 5ª, 46022 Valencia, Spain
Phone: 34 963879910. Fax: 34 963877272
E-Mail: carmonmo@iie.upv.es

<u>Response to Reviewers' comments:</u>

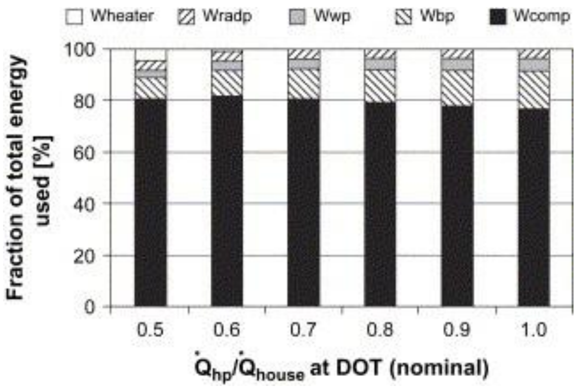
Reviewer #4:
Thank you very much for your corrections. Please find below **our answers in bold** next to your comments.

This paper presents an optimization methodology to reduce pump energy consumptions of ground source heat pump systems. Overall, this paper is well written and of a good quality. However there are still rooms for improvement. After a thorough review and careful consideration, the reviewer's recommendation is to accept this paper for publication after a minor revision according to the following comments.
* Nomenclature section needs to be improved. There are several missing nomenclatures.
Ok, thank you. It has been reviewed and improved.

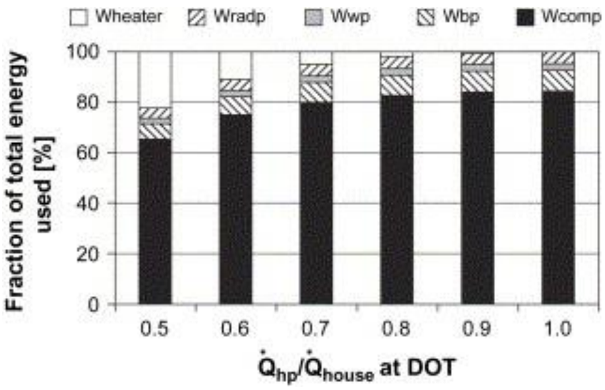
* In lines of 49-57 on Page 2, it is not clear how the undersized heat pump system with variable speed compressor control can reduce or avoid the need for supplementary heat. It should be clearly explained. The authors carefully cite someone's work to avoid plagiarism.
Ok, thank you very much for your comments.

In the work cited by the authors, a laboratory test was carried out in which two heat pumps were compared: a single speed (50 Hz) ground source heat pump and a variable speed (30-120 Hz (80 Hz nominal)) ground source heat pump. When the heat pump could not supply the capacity needed, an electric resistance (efficiency 95%) provided the extra capacity.

GSHP are normally sized to cover the peak load, but in some cases they work at part load conditions during most of the time. In those cases, and in order to avoid excessive cycling of the compressor, the heat pump can be sized to cover the base load, and use a back up heater to cover the peaks. In the case referenced by the authors, operating the compressor at a higher frequency than the nominal would mean a higher heating capacity and would reduce the need of using the electric resistance. However, the compressor efficiency would be lower than that at nominal frequency, but it would always be better than using the back up heater. The following figures show the results presented in the work referenced by the authors in which it can be observed that, for heat pumps sized to cover 55% to 60% of the load, the use of a variable-speed control provides an opportunity to reduce or to avoid the need for supplementary heat, whereas in the case of single speed control this would only be possible if the heat pump was sized to cover around 90% of the total peak load.



Variable speed - Break down of components energy requirements, respect to annual



Single speed - Break down of components energy requirements, respect to annual

This was better explained in the introduction, as requested in the review.

* It is recommended to perform a thorough editorial review. **Ok, thank you.**

We hope that now it is acceptable for publication.

In situ optimization methodology for the water circulation pumps frequency of ground source heat pump systems

C. Montagud*, J. M. Corberán, Á. Montero

Instituto de Ingeniería Energética, Universitat Politècnica de València, Camino de Vera s/n, 46022 Valencia, Spain

*Corresponding author. Tel.: +34 963879910; fax: +34 963877329. E-mail address: carmonmo@iie.upv.es (C. Montagud)

ABSTRACT

This paper presents an experimental study of the influence of the water circulation pumps frequency at the indoor and outdoor loops in the overall system performance of a ground source heat pump monitored plant, and describes a three steps new methodology to optimize the system energy performance and obtain the values of the optimal frequencies when ON/OFF regulation is employed. The first step consists of several experimental tests of pseudo-random sequence of frequency steps for both, internal and external circulation pumps, carried out during a single day. The second step is the characterisation of the quasi-steady state system performance as a function of circulation pumps frequencies. The final step consists of, by means of an analytical expression, the extrapolation of results to any partial load ratio, what allows finding the optimal frequency as a function of the instantaneous thermal load. The advantage of the proposed methodology is that it can be carried out on site and is able to consider the phenomena ocurring at the heat pump and the ground source heat exchanger when the flow rates are varied. The methodology can be applied to any installation incorporating variable speed circulation pumps. Results indicate large energy savings potential for this kind of installation.

Keywords: Heating and cooling systems, ground source heat pump, energy efficiency.

1. INTRODUCTION

Building equipments are responsible for a significant percentage of all greenhouse gas emissions produced in developed countries, being an important percentage of the total (around 60%) produced by cooling and heating systems [1-3]. According to data provided by IDAE [4], energy consumption in buildings in Spain accounts for 20% of final energy consumption, a figure that tends also to increase. In this context, the use of energy efficient and environmentally safe air conditioning systems will be desirable. Ground source heat pump systems [5-10] are recognized by the Environmental Protection Agency (EPA) among the most efficient and comfortable heating and cooling systems available today [11]. As shown in [12] this technology can lead to a 40% savings in annual electricity consumption compared to air to water conventional heat pumps.

Several studies have pointed out the considerable amount of energy consumed by auxiliary equipment in air conditioning systems [13-15]. This considerable consumption coming from auxiliary equipment is particularly relevant for ground coupled heat pump systems in which two circulation pumps are needed. Furthermore, the more efficient the heat pump is designed, the more important is to minimize the auxiliary consumption. In [16] a study is presented in which simple relations for optimum circulation rates in indirect systems are given. In [17], a theoretical study shows that there is a potential for applying an optimization method for keeping the compressor and pump speeds at the optimal combination. The key point to find this optimal combination is to consider the pump circulation rates optimizing the total power consumption of the system and not only the heat pump power consumption. The purpose of this paper is to examine the influence of the flow rate of the water circulation pumps on the overall performance of a ground source heat pump system for heating and cooling, as well as the optimization of their frequency in order to lead to the minimal energy consumption of the entire system. Research to date on control of GSHP has focussed on capacity control issues and to a lesser extent on control of secondary side working fluids. Up to the moment, still the most common type of capacity control currently deployed in commercial GSHP is ON/OFF compressor cycling. Control for ON/OFF compressor has been compared with variable speed control for a brine-to-water heat pump [18, 19]. GSHP are normally sized to cover the peak load, but in some cases they work at part load conditions during most of the time. In those cases, and in order to avoid excessive cycling of the compressor, the heat pump can be sized to cover the base load, and use a back up heater to cover the peaks. In [20-22], it was reported that this is often the case in Sweden, where heat pumps for domestic use are sized to cover 55% to 60% of the load at the design outdoor temperature which will cover 85% to 90% of the annual energy demand for heating, and in which the ability to extend the operating range of the compressor by variable-speed control provides an opportunity to reduce or to avoid the need for supplementary heat, since in the case studied the variable speed compressor could be operated between 30 and 120 Hz, being 80 Hz the nominal frequency. Operating the compressor at a higher frequency than the nominal would mean a higher heating capacity and would reduce the need of a back up heater which in the case studied was an electric resistance heater. However, the compressor efficiency would be lower than that at nominal frequency, but it would always be better than using the back up heater. Several analysis were carried out to examine GSHP system capacity and performance sensitivity to different control strategies, including control of room air temperature and water supply temperature to a fan-coils distribution circuit in an office building [23]. Following similar purposes, the development of strategies for the operation of the air conditioning ground coupled heat pump system , allowing to adapt the thermal energy generated by the system with the thermal load while satisfying the thermal comfort were studied in [24, 25]. In [24], a new management strategy is designed to diminish the consumption of the system while keeping the comfort requirements. In [25], an approach based in combining the ground source system with other production system, and decoupling energy production from energy distribution using a thermal storage device was studied. In both cases substantial energy savings, of the order of the 30%, were achieved.

In order to achieve a good system energy performance, the first step is to have a good design based on the optimal selection of components. Once the installation is designed, the second step consists in building it correctly and finally, the third step is to run the system in order to obtain a good operation. Nowadays, variable velocity circulation pumps are incorporated in most of the systems, and in any case they could be easily incorporated. This paper is focused on finding the optimal pump frecuencies for a particular ground source heat pump air conditioning facility as well as determining the way to identify this optimum by means of a simple and adequate experimental methodology. It should be emphasized that if one wants to improve the system energy performance in most of the already existing installations as well as new buildings, improving their operation is of a great importance. So, carrying out optimization strategies of this kind would result in a improvement of the building energy performance with almost no extra investments.

Nomenclature

COP	Coefficient of performance [-]
c_p	Specific heat at constant pressure [J/kg·K]

freq	Circulation pumps' frequency [Hz]
$\dot{m}$	Mass flow rate [kg/h]
PF	Performance factor [-]
$\dot{Q}$	Instantaneous thermal loads [kW]
$\dot{q}$	Building thermal load [kW]
SPF	Seasonal performance factor [-]
T	Temperature [ºC]
t	Time [s]
V	Volume of the internal hydraulic circuit [m^3]
$\dot{W}$	Power consumption [W]
Subscripts	
ambient	Refers to heat losses to the ambient
COMP	Compressor
CP	Circulation pump
db	Dead band
EC	External circuit (ground loop)
ECP	External circulation pump
el	Electrical
EXTRA	Extra time during which the ECP is switched on.
HP	Heat Pump
IC	Internal circuit (building)
ICP	Internal circulation pump
IN	Inlet
OFF	Compressor is switched off (OFF cycle period)
ON	Compressor is working (ON cycle period)
OUT	Outlet
PAR	Parasitic losses
SYSTEM	Includes heat pump, external and internal pump consumption
Greek symbols	
α	Partial load ratio [-]
Δ	Difference
η	Efficiency [-]
ρ	Density [kg/m^3]

2. GEOTHERMAL EXPERIMENTAL PLANT

Geocool plant is a reversible water to water ground source heat pump installation (15.9 kW of nominal cooling capacity and 19.3 kW of nominal heating capacity, that air-conditions a set of spaces in the Department of Applied Thermodynamics at the Universitat Politècnica de València, Spain, with a total gross flow area of approximately 250 m^2. The installation was built at the end of year 2004 and it's completely monitored (water temperature, mass flow rate and power consumption). A detailed description of the installation has been presented in previous publications (see [23] and [26] for more details). The system can be divided into two main circuits: an internal circuit which consists of a series of 12 parallel connected fan coils, an internal hydraulic loop and a water storage tank, and an external circuit which consists of the GSHX which is coupled to the heat pump by an external hydraulic loop. The ground source heat exchanger (GSHX) is made up of 6 vertical boreholes connected in a balanced parallel configuration. Each borehole has a depth of 50m and contains a single polyethylene U tube of 25 mm internal diameter, with a 70mm separation between the upward and downward tubes. The borehole overall diameter is 150 mm. The six boreholes are arranged in a 2x3 rectangular grid ($18m^2$), with a 3m separation between boreholes. All boreholes are filled with sand and finished with a bentonite layer at the top to avoid intrusion of pollutants in the aquifers. Values for ground thermal properties (conductivity of 1.43 W/mK and volumetric heat capacity of $2.25MJ/m^3K$) were obtained by means of laboratory analysis performed on soil samples although a high uncertainty (around 20%) in the estimation of the ground thermal conductivity was observed. Measurements of the ground temperature were undertaken at Geocool plant and the registered values (around 18.5ºC) were very close to the water temperature coming from the ground loop which is around 20ºC (see [26-28] for details).

A timer controls the overall system operation, which is programmed to operate between 07:00 and 22:00 h, 5 days per week. The heat pump operation is governed by an electronic controller which, depending on the building water return temperature, switches ON/OFF the heat pump compressor. The ON/OFF ground circulation pump is controlled by the heat pump controller, with a two minutes delay. In order to supply the chilled/hot water to the fan coil units, there is an internal circulation pump working continuously during the system operation. In order to vary the fan coil and GSHX water flow rates, two frequency inverters were installed; one for each circulation pump.

3. THEORY AND CALCULATION

3.1 Impact of water flow rates on the system energy performance

When optimizing the overall system performance, it is important to understand how the increase of circulating water flow rate affects the COP of the heat pump and the COP of the entire system. Figure 1 presents values of the mass flow rate and power consumption of the external circulation pump measured at different frequencies (20- 60 Hz) at the Geocool installation, being 50 Hz the nominal ones. As it is shown in Figure 1, the higher the inverter frequency, the greater the circulating water flow rate. This relationship is practically linear, as shown in the linear regression in Figure 1. However, the power consumption typically increases in a cubic manner with flow rate, as can be also appreciated in Figure 1.

On the other side, a higher water flow rate enhances the heat transfer coefficient through the heat exchanger of the heat pump and diminishes the water temperature variation across it; the same happens at the ground source heat exchanger. On the heat pump side, the increase of the water flow rate helps to reduce the temperature difference between the water and the refrigerant and, as a result, the temperature lift that must overcome the compressor becomes lower and the heat pump COP increases. These effect was analysed in [29], where a experimental campaign performed at the laboratory with a water to water heat pump of similar capacity to the one actually installed at the Geocool plant was carried out, in order to evaluate the effect of increasing the flow rate at the external circuit, while keeping constant the flow of the internal circuit, and vice versa. Results clearly led to the conclusion that the higher the flow rate at both the external and internal circuit, the better the COP of the heat pump. However, the influence is not linear and it can become difficult to assess. Manufacturers do not provide this kind of information about the influence of the secondary fluid flow rates on condenser and evaporator as, in general, they require a very precise work at the laboratory since the influences are very small and they often lie close to the uncertainty of the measurements. Regarding the ground source heat exchanger, a very low flow rate could lead to laminar flow and therefore to a poor heat transfer to the ground. This influence was checked experimentally for one day in heating mode, and results are shown in Figure 2a, where it can be observed that, for those cycles with lower water flow rate at the ground loop, corresponding to 20 Hz, the mean water temperature in the borehole heat exchanger, calculated as the average of the inlet and the outlet water temperature measurements at the ground loop when the external circulation pump is working, is slightly lower than having the system working at 50 Hz (nominal conditions). Reynolds numbers obtained at each frequency take values of 5800 at 50 Hz and 2447 at 20 Hz, which confirms that at very low frequencies of 20 Hz (low flow rates), the ground source heat exchanger is working in almost laminar conditions, which leads to a lower heat transfer coefficient and therefore, a lower water temperature. This effect will have an influence in the heat pump performance, with a reduction of the COP of the heat pump, $COP_{HP,}$ as it can be observed in Figure 2b.

Finally, regarding the fan coil units, a lower flow rate at the internal water loop means also a lower overall heat transfer coefficient and thus a lower thermal capacity of the fan coil, which could negatively affect the user comfort and turn into higher operation time and more consumption of the fan coils when the local thermal load is high [23].

So, on one hand, increasing the water flow rate on both sides (evaporator and condenser) diminishes the compressor consumption but, on the contrary, increases the circulation pumps consumption. On the other hand, reducing the water flow diminishes the circulation pumps consumption but it also increases the heat pump consumption and reduces the heat transfer coefficient in the ground source heat exchanger as well as, maybe, the user comfort. The fact that there are opposite trends on energy consumption for the variation of the circulation pumps frequency, means that there exists an optimum frequency for each one of the water loops.

It must be pointed out that in a real installation it is not possible to keep fixed the inlet water temperatures at the condenser and evaporator as it could be done at the laboratory, so, when varying the water flow rates, the consumption measurements correspond to the global effect on the whole system. However this is in fact what it is interesting, i.e. to find out which are the optimum frequencies leading to the minimum energy consumption of the whole system, and it is evident from the results that there is margin enough for important savings. The question is how to find the optimal circulation pump frequencies for a given installation. This paper presents a methodology to do so.

3.2 Evaluation of energy efficiency parameters

When a heating and cooling system is monitored under real operating conditions the first conclusion that one realizes is that the evaluation of the performance of the heat pump and the system is not easy to perform from the monitored data. The compressor is continuously cycling producing strong variations of the measured variables. In the following, the evaluation of the energy efficiency parameters from typical monitored data is presented.

First, the performance of the installation during a typical summer day will be analyzed. Figure 3 shows the water temperatures evolution at the internal and external circuits along one cooling day operation in July.

As can be seen, the water temperature sent to the outer loop (outlet of the condenser), T_{INEC} reaches a maximum value of 32°C. The ground return temperature (inlet to the condenser), T_{OUTEC}, takes values around 27 º C. This is typical in cooling season during the summer when the soil has warmed up during the month of May and June, and the daily thermal load is very high especially around midday. As can be observed, the heat pump is supplying the chilled water to the system at a temperature (outlet of the evaporator), T_{INIC}, around 7 ° C. The heat pump switches off when the return temperature T_{OUTIC} takes values around 9 º C, and switches on again when the return temperature is around 12 º C.

It can be easily observed that the compressor is cycling in such a way that, when it is switched on, there is approximately a 5K temperature difference between the inlet and outlet temperatures at both the indoor and outdoor loops approximately, (5K is the typical value employed by designers for the nominal flow rate); when the compressor turns off, both the inlet and outlet water temperatures at each circuit become the same.

The system automatically turns off from 22:00 to 07:00 h of the next day.

As shown in Figure 3, the compressor cycles present a regular evolution along the day except for the first one that has a longer duration. This is because water at the internal circuit has heated up during the night, reaching a stationary value of around 23ºC at 7 am when the system switches on, and it takes time to the compressor to cool down the water of the circuit again to the required operating temperature. It can also be noticed in Figure 3 that the water temperature coming from the ground loop, T_{OUTEC}, increases along the day. The main reason for this is that the water temperature coming from the ground is heated up as the heat is injected into the ground along the day. Some variation in the cycles' duration can be distinguished at midday from 13:00 to 17:00 which correspond to higher ambient temperatures and solar irradiation, causing a consequently higher thermal demand in the building. Analogue to this, the length of the cycles is lower in the evening after 19:00h due to a decrease in the building thermal demand. Nevertheless, except for the first cycle in the morning, the duration of the cycles is approximately the same and the daily performance could be characterized by a single ON/OFF compressor cycle that stood for the mean daily thermal demand considering mean water temperatures at the internal and external circuits. So, in order to characterize the typical performance of the system, a characteristic period ON/OFF could be analyzed, and the energy performance factor of the day would correspond to the performance factor of one characteristic cycle ON/OFF.

Energy efficiency parameters are calculated from the power consumption readings and the heat transferred from the heat pump to the internal circuit, which is estimated from the values of T_{INIC}, T_{OUTIC} and the water flow rate $\dot{m}$ (measured with four wire PT100 temperature sensors and a Coriolis meter). Instantaneous thermal loads are obtained by the energy balance from inlet to outlet of the internal heat exchanger. Expression 1 shows an example for cooling mode operation.

$$\dot{Q}_{HP}(t) = \dot{m}c_p\left(T_{OUTIC}(t) - T_{INIC}(t)\right) \qquad (1)$$

Figure 4 shows the evolution of the thermal load and the power consumption of each system component, except for the fan coil units, for a single ON/OFF cycle of the heat pump in July at 12:30 pm (zoom of Figure 3). As can be observed in Figure 4, the thermal load is quite high as the duration of the ON period is much greater (22 minutes) than the OFF period (8 minutes). So, the partial load ratio defined as the relationship between the duration of the ON period and the total time for each cycle of the compressor will take a value around 0.7. The discontinuous line stands for the instantaneous measurements whereas the continuous line stands for the mean values calculated for the cycle. The instantaneous heat transferred in the evaporator (internal circuit) is calculated using Equation (1), and the power consumption of each individual component, i.e., the internal circulation pump, the external circulation pump and the heat pump compressor unit, is measured and collected every minute in the data acquisition system. The total average consumption of the system, which takes a mean value of 3.62kW, has been calculated as the addition of the circulation pumps consumption, the compressor power and the heat pump parasitic losses during the ON period. It can be observed in Figure 4 that the compressor power gets the greatest value and stands for the 83% of the total consumption. It can also be noticed that, although the internal circulation pump consumption is much lower than the compressor one, it is working all the time and not only during the ON period, standing for a 9.94% of the total electricity consumption, whereas the external circulation pump is cycling with the compressor but delayed two extra minutes in time. With respect to the heat pump parasitic losses, they were measured several times when the compressor is switched off by means of a wattmeter, and turned to have a value around 60W, which is the 1.66% of the total consumption but it should be pointed out that they are present during all the operational time of the unit. The power consumption of the internal circulation pump is fairly high, getting values around 12% of the compressor power. This is especially relevant when trying to optimize the whole system performance and its influence will be analysed in the present work.

The integral of Equation (1) during the ON period stands for the total heat transferred to the evaporator during the cycle (on and off time), which will compensate the thermal load of the building. Likewise, the system energy consumption is calculated by numerically integrating the power consumption measured for each of the system components: internal circulation pump, external circulation pump, heat pump and fan coil units.

Energy efficiency is characterized by the energy performance factor, defined as the ratio between the thermal load and the electric energy consumption during a time interval. Depending on the duration of the integration period, the energy performance factor can be seasonal, monthly, daily, etc. The most representative one is the seasonal performance factor (SPF) that estimates the system performance during each season (winter or summer).

When calculating the performance factor for a single ON/OFF cycle, if only the ON time is considered, the performance factor integrated for one cycle will be the same as the coefficient of performance at steady state conditions. In fact, it was proven in [30] that water to water units have negligible start up losses, so that partialization losses only depend on the parasitic losses due to the electronics (or maybe the crankcase heater consumption). During the ON period, the unit works as in steady state conditions with condensing and evaporating temperatures gliding with the respective inlet water temperatures variation.

The system coefficient of performance will be calculated using expression (2) during the ON period for each cycle, and therefore, it will correspond to the performance of the unit and the system under quasi-steady state conditions.

$$COP_{SYSTEM} = \frac{\int_{0}^{t_{ON}} \left(\dot{Q}_{HP}(t) \pm \eta_{el} \cdot \dot{W}_{ICP}(t) \right) \cdot dt}{\int_{0}^{t_{ON}} \left(\dot{W}_{COMP}(t) + \dot{W}_{PAR}(t) + \dot{W}_{ECP}(t) + \dot{W}_{ICP}(t) \right) dt} \qquad (2)$$

As it is indicated in expression (2), the system coefficient of performance COP_{SYSTEM} includes all components required to provide the hot or chilled water at the inlet of the terminal devices: heat pump compressor and parasitic losses, and both the external and internal circulation pumps consumption.

The water temperature sensors are placed right at the inlet and outlet of the heat pump, so that the instantaneous heating/cooling capacity of the heat pump can be estimated by Equation (1). When analyzing the system coefficient of performance, COP_{SYSTEM}, which includes the internal circulation pump consumption, the heating/cooling capacity that should be taken into account is the one transferred to the building. As a matter of fact, the internal circulation pump losses contribute to heating up the fluid; for this reason, the internal circulation pump consumption needs to be added to the heat pump capacity during heating mode while needs to be extracted during cooling mode. That is the reason for the ± sign as shown in the numerator of Equation (2).

However, not all the electrical power consumed by the circulation pump is transferred to the fluid, since part of these losses go to the surroundings, mainly the electrical losses generated at the motor which are transferred to the air through the motor envelope and the fan. Considering the definition of the electrical efficiency of the pump expressed in Equation (3), the extra heat supplied to the water can be estimated by multiplying the internal circulation pump consumption by this efficiency, η_{el} as shown in Equation (2).

$$\eta_{el} = 1 - \frac{\dot{Q}_{ambient}}{\dot{W}_{ICP}} \quad (3)$$

Where $\dot{Q}_{ambient}$ represents the heat losses of the circulation pump electric motor, transferred to the ambient. The electric power transferred to the shaft of the pump rotor corresponds to the difference between the internal circulation pump electric consumption $\dot{W}_{ICP}$ and the heat losses to the ambient $\dot{Q}_{ambient}$. Finally, this will result in an added heat transferred to the fluid, resulting in an increase of its temperature.

For the type of motor corresponding to the circulation pumps at Geocool plant (Grundfos MG71, IP54, F, 3 x 220-240D V, 2.3 A) [31], typical values of electrical efficiency around 85-90% for a nominal frequency (50 Hz) should be considered. These values would vary with the frequency; however, it is very difficult to get this information from the manufacturer. In the study, a constant efficiency value of 90% has been assumed. Nevertheless, the influence of this parameter on the results of the proposed methodology is almost negligible comparing order of magnitudes of the compressor and pump consumptions.

Finally, considering both the ON and OFF time for the integration period of Equation (2), the calculation for the performance factor of the system along one entire representative cycle would read as follows:

$$PF_{SYSTEM} = \frac{\int_0^{t_{OFF}+t_{ON}} \left(\dot{Q}_{HP}(t) \pm \eta_{el} \cdot \dot{W}_{ICP}(t)\right) \cdot dt}{\int_0^{t_{OFF}+t_{ON}} \left(\dot{W}_{COMP}(t) + \dot{W}_{PAR}(t) + \dot{W}_{ECP}(t) + \dot{W}_{ICP}(t)\right) dt} = \tag{4}$$

$$= \frac{\dot{Q}_{HP} \cdot t_{ON} \pm \eta_{el} \cdot \dot{W}_{ICP} \cdot (t_{ON} + t_{OFF})}{\left(\dot{W}_{COMP} + \dot{W}_{ECP} + \dot{W}_{ICP} + \dot{W}_{PAR}\right) \cdot t_{ON} + \dot{W}_{ICP} \cdot t_{OFF} + \dot{W}_{ECP} \cdot t_{EXTRA} + \dot{W}_{PAR} \cdot t_{OFF}}$$

Where t_{EXTRA} is the extra time during which the external circulation pump is switched ON with respect to the compressor. In the case of Geocool installation, the external circulation pump switches on one minute before the compressor and turns off one minute later, so t_{EXTRA} takes a value of 2 minutes. This is typical in this kind of installation. The circulation pump is always switched on before the heat pump in order to activate the flow switch. And the operation is also normally repeated after the heat pump is switched off although the reason for this is not entirely clear in opinion of the authors.

If t_{ON} and t_{OFF} are representative of the average cycle of the day, then the defined performance factor PF_{SYSTEM} will become the daily performance factor of the system. The selection of a representative value for t_{ON} and t_{OFF} can be easily done just by expressing them as a function of the average load factor for the day as will be described later on.

4. OPTIMIZATION: PROPOSED METHODOLOGY

Optimization studies were undertaken using a quasi-steady state mathematical model of the installation developed in EES [32] which had been previously validated by comparison with experimental data. The results of these studies were presented in [23]. Variables analysed included internal circuit mass flow rate, set-point temperature and temperature bandwidth of the heat pump control, building space set-point temperature and building space temperature bandwidth. The study gave clear indication of the existence of optimum frequencies for the circulation pumps depending on the operating conditions, which in general agree well with the results obtained in later experimental studies.

However, even with the most sophisticated and detailed model, it is not possible to take into consideration the wide ensemble of parameters which in practice affects the performance of a geothermal installation, as for instance, the user's daily activity, the ground thermal response, ambient temperature variations along the day...Therefore, it was decided to try to develop a methodology to get the optimum frequencies in situ under real operating conditions. After several trials, a new methodology for the in situ optimization of the frequency of each of the water circulation pumps of a geothermal system with ON/OFF regulation was developed and has been successfully tested along more than one year of operation allowing an accurate definition of the optimal frequencies for the circulation pumps of the system.

The proposed methodology consists of the following 3 steps:

Step 1: ON time operation characterisation

The objective of this step is the characterisation of the system performance during the ON time of the compressor at different frequencies for the internal and external circulation pumps. It consists of monitoring the consumptions and performance of the system along one day at different values of the frequency of the circulation pumps. In order to account for the possible influence of the ground thermal response variation along the day, a random sequence of frequency variation is proposed. Figure 5 shows an example of the possible test sequence for a variation of the frequency of the circulation pumps from 20 Hz to 60 Hz.

When the compressor is ON, the control of the unit makes the return water temperature of the internal circuit to the heat pump stay always under the desired range, so that the thermal conditions at the heat pump do not depend on the external ambient temperature or user needs. The only exeption to this is the first heating/cooling cycle of the day which in fact is dedicated to bring the water of the internal circuit to the desired temperature range. Therefore, the proposed tests must be always carried out after this first initializing cycle has been performed and the system is at normal operating conditions. In any case, as some initialization also takes place around the boreholes, it is recommended to start with the tests after at least one hour of normal operation.

As the obtained results may be influenced by the average ground temperature, and this changes along the year, as well as with the compensation of the setting temperature for the water return temperature, it is recommended to perform the tests during 4 days a year: two during the heating season and two during the cooling season, for example, one test at the middle of the season and one at the end of the season. In the Geocool installation the optimum frequencies obtained from different days along one season are quite the same [33]. The reason to propose to repeat the test along one season is exactly to assess the influence of the ground temperature, and confirm if the optimum stays at the same point, as it happens in Geocool installation, or if it requires some readjustment along the season.

Figure 5 shows an example of the strategy followed for a test along one day corresponding to heating mode: the continuous line stands for the water flow rate at the internal hydraulic loop, whereas the discontinuous line stands for the water flow rate at the external loop. As can be observed in Figure 5, the cycles of the heat pump are well defined by the evolution of the external circuit flow rate, as the ON/OFF periods of the heat pump match the exterior circulation pump operation. The circulation pump frequencies are varied for each ON cycle. Once the test is finished, the coefficient of performance of the system COP_{SYSTEM} can be evaluated for each ON cycle so that it will allow the construction of the performance maps of the unit as a function of both the external and internal circulation pumps frequency.

Based on the authors' experience, in order to properly measure and estimate the performance along each ON cycle, the time ON per cycle should be around ten minutes or higher. That means that for the day when the test is planned to be carried out, the thermal load of the building needs to be manipulated in order to have cycles with at least 10 minutes ON time duration. This is easy to adjust in situ by changing the number of active and fan speed of the fan coils and even opening or closing windows provided that the building is not in normal use. This is the reason why it is desirable to carry out the tests during non-working days.

Step 2: Estimation of the system COP maps

Step 2 consists of the analysis of the results obtained from step 1 and the estimation of the COP_{SYSTEM}, as defined by eq. (2) as a function of the circulation pump frequencies. Figure 6 shows the COP_{SYSTEM} maps obtained for one typical cooling and one typical heating day in the Geocool installation.

Figures 6a and 6b show that the optimum COP_{SYSTEM} corresponds to low pump frequencies. The maximum COP_{SYSTEM} value corresponds to an internal frequency of around 40 Hz and an external frequency of around 30 Hz for heating mode as shown in Figure 6a. In the case of cooling mode operation, the optimal pair of frequencies results approximately in 28Hz for the internal circulation pump, and around 40Hz for the external circulation pump. It can be observed that the optimal frequency for the internal circulation pump results in a higher value for heating mode operation. This is because the internal circulation pump consumption contributes to the heating adding a useful heat, and the other way around in cooling mode. The optimal pair of frequencies obtained from Figure 6 corresponds to quasi-steady state working conditions. In practice, steady state will only happen when the building thermal load equals the heat pump capacity, what should never happen since the heat pump is designed in order to be able to satisfy the maximum peak load for heating and cooling mode and the nominal capacity is always greater than the building load.

Step 3: Estimation of the system performance maps for any thermal load

The quasi-steady performance maps of the unit were obtained during the ON time duration of the compressor. However, as previously analysed in the present work, during the OFF time period, there is a power consumption that can significantly degrade the daily performance factor of the system. In this 3rd step, it will be possible to account for this influence and calculate, from the quasi-steady performance maps obtained in the 2nd step of the methodology, the optimal frequencies as a function of the partial load ratio. The ratio between the total thermal load that the system copes with and the heat pump capacity is known as the load ratio or 'partial load ratio' and can be evaluated by the following expression:

$$\alpha = \frac{\dot{q} \pm \eta_{el} \cdot \dot{W}_{ICP}}{\dot{Q}_{HP}} \tag{5}$$

Where the heat generated by the internal circulation pump must be considered continuously since it is always kept ON, and must be added to the building thermal load during the cooling season and subtracted during the winter season (heats are considered in absolute value so they have the same sign for heating and cooling).

When ON/OFF regulation is employed, the ratio between the total thermal load to the system and the capacity of the heat pump results in the end in the cycling of the compressor, and the partial load ratio can be evaluated as the relationship between the ON time of the compressor and the total time duration for each cycle as expressed in Equation (6):

$$\alpha = \frac{t_{ON}}{t_{ON} + t_{OFF}} \tag{6}$$

Where t_{ON} and t_{OFF} are the ON and OFF operational time for one cycle respectively.

Given a nominal heat pump capacity and a nominal internal circulation pump consumption, the number of cycles of the day strongly depends on the building thermal load. As already shown in Figure 4, the external circulation pump operates an extra time for each cycle and the internal circulation pump must operate continuously throughout the day, which actually makes the optimum strongly dependent on the number of cycles and the relationship between the ON and the OFF periods duration, i.e. the partial load ratio. Fortunately, a simple analytical methodology can be employed to obtain the system performance factor PF_{SYSTEM} as a function of the partial load ratio as it will be described in the following.

The role of the tank in the internal loop is to increase its thermal inertia (which will depend on the tank volume as well as on the internal volume of the piping work and terminal units) in order to reach adequate durations for the ON/OFF periods. The dynamics of the distribution system of a water to water GSHP can be basically described by the energy conservation equation applied to the total volume of water *V* contained in the distribution system. If $\dot{Q}_{HP}$ is the cooling/heating capacity of the heat pump, $\dot{W}_{ICP}$ is the internal circulation pump consumption and $\dot{q}$ is the thermal load of the building, taking into account that the ON/OFF cycle is controlled by the thermostat differential (deadband), ΔT_{db}, and assuming that, along the analysed period, both $\dot{Q}_{HP}$ and $\dot{q}$ remain constant as well as the internal circulation pump consumption $\dot{W}_{ICP}$, the time duration corresponding to the ON and OFF periods results:

$$t_{ON} = \left(\rho V c_p \Delta T_{db}\right) \Big/ \left(\dot{Q}_{HP} - \left(\dot{q} \pm \eta_{el} \cdot \dot{W}_{ICP}\right)\right) \qquad t_{OFF} = \left(\rho V c_p \Delta T_{db}\right) \Big/ \left(\dot{q} \pm \eta_{el} \cdot \dot{W}_{ICP}\right) \tag{7}$$

Therefore, the ON time can be expressed in terms of the water volume of the tank and the pipes of the internal hydraulic circuit, *V*, the partial load factor, α, and the dead band temperature of the heat pump controller, ΔT_{db}, as represented on Equation (8).

$$t_{ON} = \frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{\dot{Q}_{HP}\left(1-\alpha\right)} \tag{8}$$

Using the definition of the partial load ratio (6) and Equation (8), it is possible to obtain the system performance factor as a function of the coefficient of performance of the system COP_{SYSTEM} at quasi-steady state conditions, at different partial load ratios α expressed by Equation (9).

$$PF_{SYSTEM} = \frac{COP_{SYSTEM} \pm \frac{(1-\alpha)}{\alpha} \cdot \frac{\eta_{el} \cdot \dot{W}_{ICP}}{\sum \dot{W}}}{1+\left[COP_{SYSTEM} \mp \frac{\eta_{el} \cdot \dot{W}_{ICP}}{\sum \dot{W}}\right] \cdot \frac{\dot{W}_{ECP} \cdot (1-\alpha)}{V \cdot \rho \cdot c_p \cdot \Delta T_{db}} \cdot t_{EXTRA} + \frac{(1-\alpha)}{\alpha} \cdot \frac{\left(\dot{W}_{ICP} + \dot{W}_{PAR}\right)}{\sum \dot{W}}} \quad (9)$$

Where $\sum \dot{W}$ stands for the total system consumption including both the internal and external circulation pumps, the compressor consumption and the heat pump electrical parasitic losses consumption, and COP_{SYSTEM} is obtained in step 2, as defined in (2).

The experimental information needed in Equation (9) comes from the first step (experimental measurements varying both the internal and external circulation pumps) and the second step where the performance maps of the system at quasi-steady state conditions are built. Looking at Equation (9) it can be noticed that when the partial load ratio α equals to one, the system performance factor PF_{SYSTEM} takes the same value as in quasi-steady state conditions, COP_{SYSTEM}.

The third step of the proposed methodology would therefore consist in using Equation (9) to extrapolate the performance maps characterisation at quasi-steady state conditions obtained in the 2nd step to any partial load ratio. Results are shown in Figures 7 and 8 for heating and cooling mode respectively.

It can be observed from Figures 7 and 8 that, low partial loads degrade the performance of the system leading to lower optimal frequencies, because the auxiliaries consumption (external and internal circulation pumps, and parasitic losses) have a great influence in the system performance factor, as it was predicted by Equation (9). On the contrary, the higher the partial load factor, the more similar the optimal frequencies are to the ones corresponding to quasi-steady state conditions represented in Figure 6, because the influence of the auxiliaries consumption turn out to be practically negligible.

In the above, it has been assumed that when the compressor switches ON the consumption and the capacity of the heat pump becomes equal to the values corresponding to quasi-steady conditions. This assumption could not be valid for air source heat pumps; but for ground source heat pumps it is very close to reality as concluded in detailed measurements performed at laboratory conditions where it was found that the partialization losses of this kind of heat pumps are negligible [30].

Finally, it is important to highlight that the proposed methodology has been designed in such a way that it can be carried out with minimum monitoring instrumentation. The whole procedure only requires the measurement of the electrical consumption of the circulation pumps and its frequency, the electrical consumption of the heat pump, and the evaluation of the heat pump capacity at the internal circuit through the measurement of the inlet ant outlet temperatures and the flow rate. In case that flow rates are not measured, it becomes necessary to perform the characterisation of the hydraulic behaviour of the system and get an estimation of the relation between flow rate and frequency. In order to take into account the effect of the temperature on the variation of the water properties in the loops, especially at the internal one, it is recommended to perform the characterisation tests once for winter and once for summer. This can be perfectly done in conjunction with the test campaign of step 1 of the proposed methodology, leading to a better accuracy in the determination of the optimal frequencies.

It should be stressed that the expressions developed on the third step of the proposed methodology are valid for a water to water reversible heat pump working with a single ON/OFF compressor. Further studies are to be developed to extend the methodology for tandem or variable speed compressors.

5. RESULTS AND DISCUSSION: OPTIMAL FREQUENCIES

Table 1 presents a summary of the optimal frequencies for the internal and external circulation pumps at different partial load ratios for heating and cooling mode operation. Notice that the summit of the performance factor maps are quite flat, therefore the selected values indicate more a good range for the frequency, rather than an optimal absolute value.

It can be concluded from Table 1 that, for those periods such us spring and autumn when the building energy demand is expected to be low, it is to say, the partial load ratio will be lower than 0.2 or even 0.1, the system should be running at lower circulation pump frequencies reaching an optimum at 20 Hz for the internal circulation pump and 40 Hz for the external circulation pump at both cooling and heating mode. For those other periods when the installation is expected to be running at very high load factors at extreme summer and winter conditions, the optimal frequencies result in 30Hz for the internal circulation pump and 40Hz for the external circulation pump in cooling mode; whereas for heating mode, the optimal frequencies are 40Hz for the internal circulation pump and 30Hz for the external circulation pump.

Further optimization analysis were developed at different days for both cooling and heating mode, so that it was possible to compare results and verify the optimal frequency values that have been presented in this paper. Obtained results led to the same optimal frequencies for heating and cooling mode with an uncertainty of ±5Hz. Anyway, as commented above, this methodology does not intend to define exact values for the optimal frequencies, but just a reasonable good estimation of the frequencies for each load ratio.

Analytical expressions could be developed from the results shown in Table 1 for the optimal frequencies as a function of the load ratio and could be programmed in the control board of the system, so that optimization control algorithms could be used to make the system run at the optimal frequency depending on the building thermal energy demand, which for sure would lead to significant energy savings along the year.

An estimation of the potential energy savings was carried out for each month of the year. Results are shown in Figure 9, where the total energy consumption measured for each month is compared to the optimal values obtained in case that the optimisation methodology were applied. As shown in Figure 9, a maximum value of 37% energy savings could be obtained on November where the heating thermal load is minimal in Valencia, being the total annual energy savings of 28% compared to nominal circulation pump frequencies (50Hz for both the external and the internal hydraulic loops).

Finally, it should be pointed out that running at low frequencies around 20 Hz in cooling mode, could cause the outlet water temperature at the evaporator of the heat pump be too low, so that an anti-frost alarm should be programmed in the control of the system to protect the unit from freezing. Additionally, running at low frequencies, could negatively affect user comfort at the air conditioned spaces. Therefore, the above mentioned savings would be an indication of the maximum potential energy savings provided that the control system will have to always warrant user comfort.

6. CONCLUSIONS

The analysis and energy optimization of the circulation pump frequencies for a GSHP system in an institutional building at the Universitat Politècnica de València was carried out. Results obtained in this research work demonstrate that there exists a clear trend of system response to changes in the frequencies of the circulation pumps. After several trials, it was concluded that a performance characterisation of the system needs to be done in situ and a new methodology for optimization under quasi-steady conditions was found which is able to lead to a clear identification of the optimal frequencies at any building thermal load for heating or cooling season.

The proposed methodology is basically composed of three steps. The first step consists in the experimental characterisation of the system energy performance (COP_{SYSTEM}) as a function of the circulation pump frequencies during one entire day for heating and one for cooling. The second step consists in the quasi-steady state performance maps construction. Finally, the third step allows extrapolating the quasi-steady COP_{SYSTEM} values, obtained at the 2nd step of the proposed methodology, to any load ratio. Therefore, the optimum frequencies at any load ratio can be found.

The methodology has been applied at different days for both heating and cooling seasons, and results obtained led to the following conclusions:

- Results for quasi-steady state conditions showed that there is a different optimum in cooling and heating mode.
    - In heating mode, the optimal circulation pump frequencies result in approximately 40 Hz for the internal pump and 30 Hz for the external one.
    - In cooling mode, the optimal circulation pump frequencies result in approximately 30 Hz for the internal pump and 40 Hz for the external one.
- The performance factor of the system degrades with the partial load ratio leading to lower optimal circulation pump frequencies at lower loads. For those periods when the building energy demand is expected to be low (spring or autumn), and the partial load ratio takes values lower than 0.2 or even 0.1, the optimal frequencies result in 20 Hz for the internal circulation pump and 40 Hz for the external one at both cooling and heating mode.

The aim of the proposed optimization methodology is to use the system performance maps obtained, in order to run the system at its optimal operating point at any load ratio. The results show that there is a high potential for energy savings which has been estimated as 28% of annual electricity consumption for the studied case.

Further optimization is to be carried out in Geocool installation where appropriate control algorithms, including adequate control strategies to ensure users comfort will be implemented in order to optimize the daily energy performance of the system.

ACKNOWLEDGEMENTS

This work was supported by the European Commission under the FP7 framework project GROUND-MED: “Advanced ground source heat pump systems for heating and cooling in Mediterranean climate”.

REFERENCES

[1] M. Orme, Estimates of the energy impact of the ventilation and associated financial experience, Energy and Buildings, 33 (2001) 199-255.

[2] International Energy Agency (IEA), in: http://www.iea.org/papers/2011/buildings_roadmap.pdf, April 2012.

[3] P. Champsour, INSEE (Institut de Statistique et des Etudes Economiques), Paris, France, Recensement de la Population 1999 – Des logements Plus Grands et Plus Comfortables, 2000, p. 4 ISSN 0997-3192.

[4] Instituto para la Diversificación y Ahorro de la Energía (IDAE), in http://www.idae.es/index.php/idpag.23/relmenu.344/mod.pags/mem.detalle, April 2012.

[5] J.W. Lund, D.H. Freeston, World-wide direct uses of geothermal energy 2000, Geothermics, 30 (2001) 29-68.

[6] J.W. Lund, Geothermal heat pumps – an overview, Quarterly Bull, Geo-Heat Centre, 22(1) (2001) 1-2.

[7] B. Sanner, C. Karytsas, D. Mendrinos, L. Rybach, Current status of ground source heat pumps and underground thermal storage in Europe, Geothermics 32 (2003) 579-588

[8] A.M. Omer, Ground-source heat pump systems and applications, Renewable and Sustainable Energy Reviews 12 (2008) 344-371.

[9] J.D. Spitler, Ground-Source Heat Pump System Research - Past, Present and Future, HVAC&Research 11 (2) (2005).

[10] K.J. Chua, S.K. Chou, W.M. Yang, Advances in heat pump systems: A review, Applied Energy 87 (2010) 3611-3624.

[11] Energy Star Program from US Environment Protection Agency, in: http://www.energystar.gov/index.cfm?fuseaction=find_a_product.showProductGroup&pgw_code=HP, April 2012.

[12] J.F. Urchueguía, M. Zacarés, J.M. Corberán, Á. Montero, J. Martos, H. Witte, Comparison between the energy performance of a group coupled water to water heat pump system and an air to water heat pump system for heating and cooling in typical conditions of the European Mediterranean Coast, Energy Conversion and Management 49 (2008) 2917-2923.

[13] M. Bernier, B. Bourret, Pumping Energy and Variable frequency drives, ASHRAE Journal (December 1999), 37-40.

[14] J. R. Brodrick, D. Westphalen, Uncovering Auxiliary Energy Use, ASHRAE Journal (February 2001), 58-61.

[15] W. P. Banhfleth, E. Peyer, Variable primary flow chilled water systems: potential benefits and application issues, report ARTI-21CR/611-20070-01 prepared for the Air-Conditioning and Refrigeration Technology Institute, Arlington, Virginia, US.

[16] E. Granryd, Optimum circulation rates in indirect systems, Proceedings of the IIR Conference on Minimum Charge Refrigeration Systems, Stockholm, Sweden, 2002.

[17] F. Karlsson, Capacity control of Residential Heat Pump Heating Systems, PhD Thesis, Department of Energy and Environment, Chalmers University of Technology, Göteborg, Sweden (2007).

[18] P. Fahlén, F. Karlsson, Improving efficiency of hydronic heat pump heating systems, 21st International Congress of Refrigeration, Washington, USA. International Institute of Refrigeration (ICR 0550, 2003).

[19] Fahlén P,Karlsson F, Optimizing And Controlling Media Flows In Heat Pump Systems, 8th IEA Heat Pump Conference, Las Vegas, USA, 2005.

[20] F. Karlsson, P. Fahlen, Capacity-controlled ground source heat pumps in hydronic heating systems, International Journal of Refrigeration 30 (2007) 221-229.

[21] F. Karlsson, P. Fahlen, Impact of design and thermal inertia on the energy saving potential of capacity controlled heat pump heating systems, International Journal of Refrigeration 31 (2008) 1094-1103.

[22] P. Fahlen, Efficiency aspects of heat pump heating systems – load matching and parasitic losses, IEA Heat Pump Centre Newsletter, Volume 26 – No. 3/2008, 38-48.

[23] J.M. Corberán, D.P. Finn, C. Montagud, F.T. Murphy, K.C. Edwards, A quasi-steady state mathematical model of an integrated ground source heat pump for building space control, Energy and Buildings 43 (2011) 82-92.

[24] N. Pardo, Á. Montero, A. Sala, J. Martos, J. F. Urchueguía, Efficiency Improvement of a ground coupled heat pump system from energy management, Applied Thermal Engineering 31 (2011) 391-398.

[25] N. Pardo, Á. Montero, J. Martos, J. F. Urchueguía, Optimization of hybrid –ground coupled and air source- heat pump systems in combination with thermal storage, Applied Thermal Engineering 30 (2010) 1073-1077.

[26] C. Montagud, J.M Corberán, Á. Montero, J.F. Urchueguía, Analysis of the energy performance of a Ground Source Heat Pump system after five years of operation. Energy & Buildings 43 (2011), 3618-3626.

[27] T. Magraner, Validación Experimental de los Métodos de Diseño de Instalaciones de Bomba de Calor Acoplada al Terreno. PhD Thesis, Universitat Politècnica de València, Valencia (Spain) (2010).

[28] T. Bandos, Á. Montero, E. Fernández and J.F. Urchueguía, Improving parameter estimates obtained from thermal response tests: Effect of ambient air temperature variations. Geothermics 40 (2011) 136-143.

[29] J.M. Corberán, C. Radulescu, J. Gonzalvez Macia, Performance characterisation of a reversible water to water heat pump. 9th IEA Heat Pump Conference, Zurich, (2008).

[30] J.M. Corberán , D. Donadello, I. Martínez-Galván, C.Montagud, Partialization losses of on/off operation of water-to-water refrigeration/heat pump units. ICR, Prague (Czech Republic) (2011) ID 720.

[31] Available on http://www.grundfos.com/products/, May 2012.

[32] EES 2008. Engineering Equation Solver, F-Chart Software. The University of Wisconsin. Madison.

[33] J.M. Corberán, C. Montagud, F. Heselhaus, Energy optimization of a ground source heat pump system for heating and cooling in an office building, ISHPC11 Sources/Sinks alternative to the outside Air for Heat Pump and Air-Conditioning Techniques (Alternative Sources - AS), IIR International Conferences, Padua(Italy) (2011) 93–102.

List of Figure Captions

Figure 1 - Power consumption and water flow rate as a function of the pump frequency.

Figure 2 – Effect of varying the water flow rate on the heat transfer at the borehole heat exchanger: a) Effect on the outlet water temperature b) Effect on the heat pump COP.

Figure 3 – Daily temperature evolution at the internal and external circuits for a cooling day.

Figure 4 – Power consumption of the system components, for an ON/OFF cycle of the heat pump.

Figure 5 - Frequency variation strategy in heating mode.

Figure 6 - COP_{SYSTEM} quasi-steady state maps as a function of pump frequency a) Heating mode b) Cooling mode.

Figure 7- PF_{SYSTEM} maps as a function of pump frequency at heating mode: a) α =0.8, b) α =0.5,c) α =0.2, d) α =0.1.

Figure 8 - PF_{SYSTEM} maps as a function of pump frequency at cooling mode: a) α =0.8, b) α =0.5,c) α =0.2,d) α =0.1 .

Figure 9 – Energy savings estimation for one year of operation.

Table(s) with Caption(s)

		α =1	α =0.8	α =0.5	α =0.2	α =0.1
ICP frequency (Hz)	Heating mode	40	40	37.5	27	22
	Cooling mode	30	30	25	22	20
ECP frequency (Hz)	Heating mode	30	30	32.5	40	40
	Cooling mode	40	40	40	40	40

Table 1 - Optimal frequencies for heating and cooling mode operation at different partial load factors.

Figure(s)

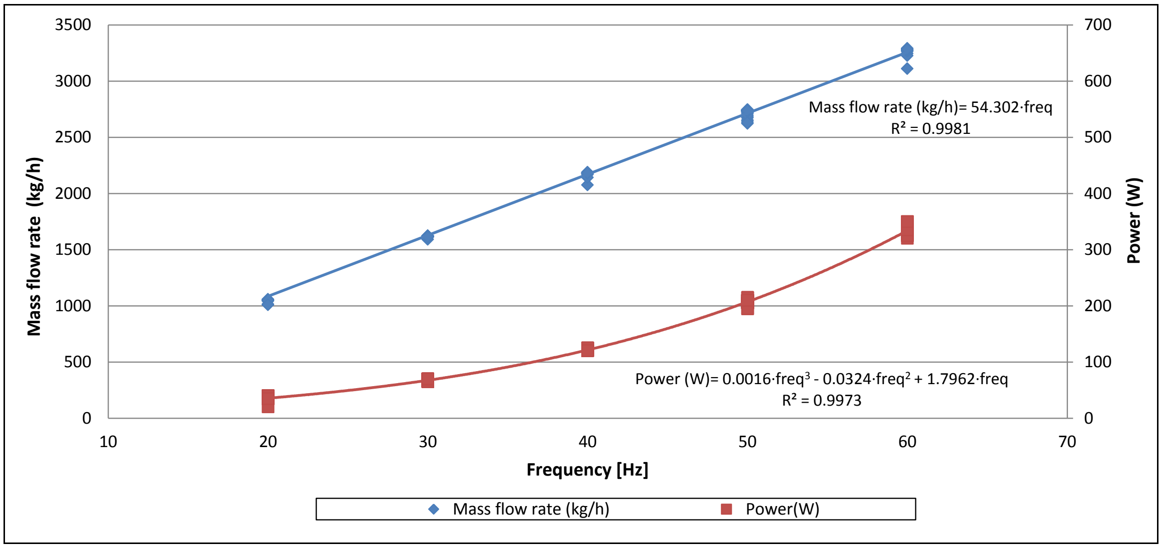
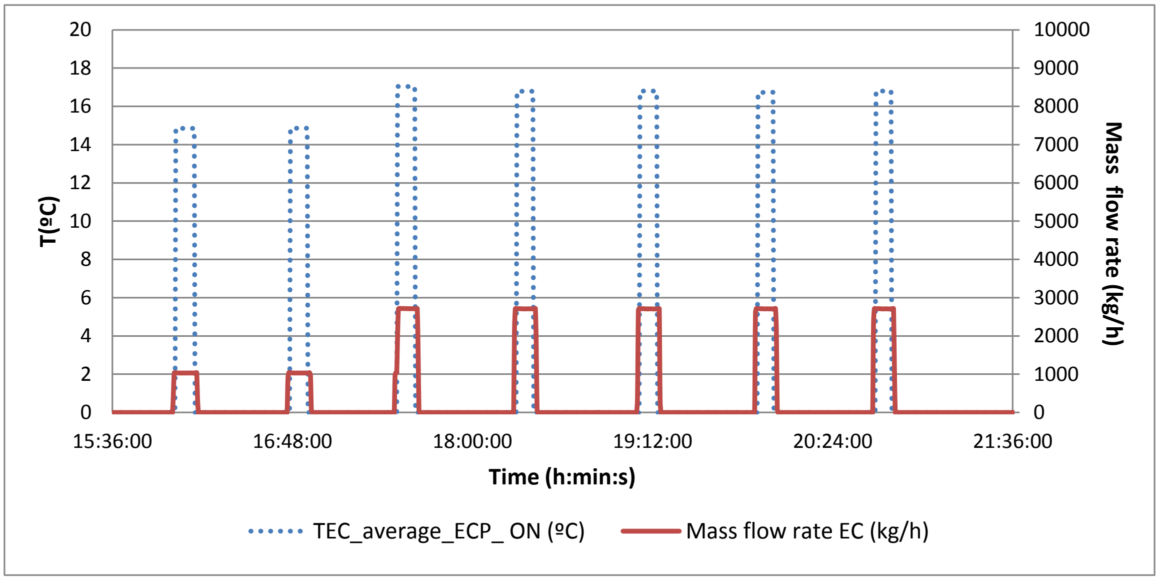


Figure 1 - Power consumption and water flow rate as a function of the pump frequency.

a)



b)

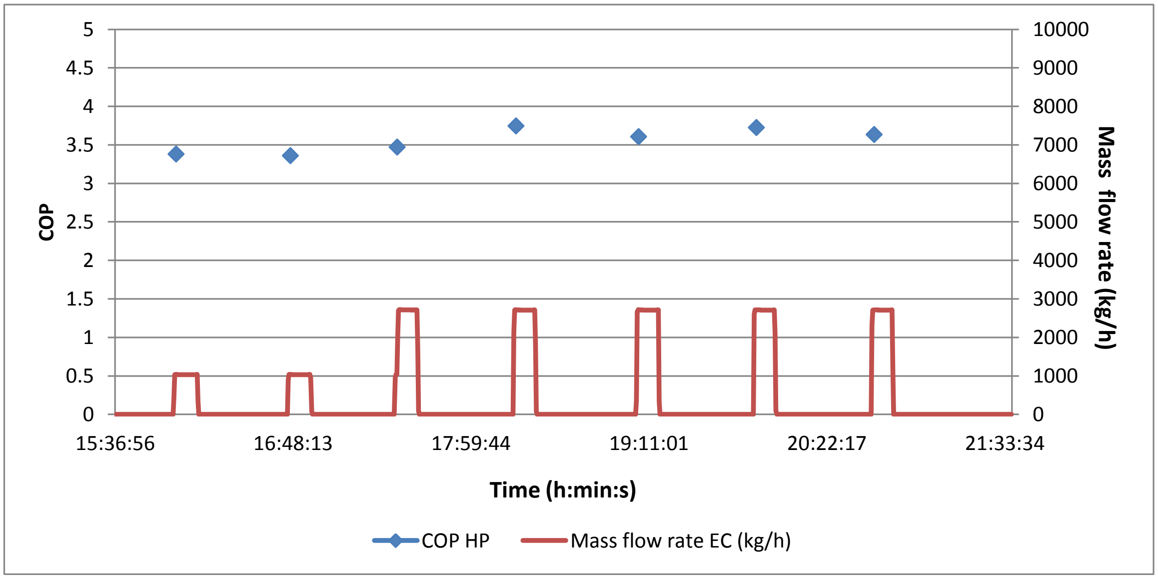


Figure 2 – Effect of varying the water flow rate on the heat transfer at the borehole heat exchanger: a) Effect on the outlet water temperature b) Effect on the heat pump COP.

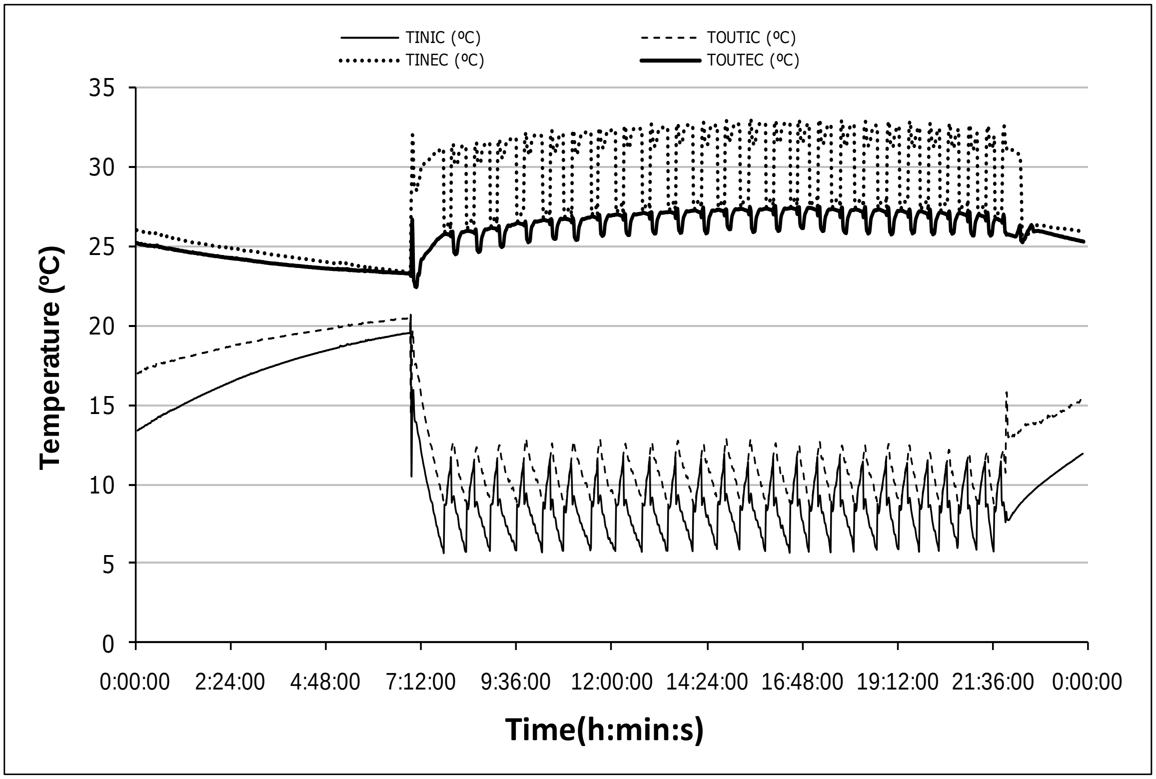


Figure 3 – Daily temperature evolution at the internal and external circuits for a cooling day.

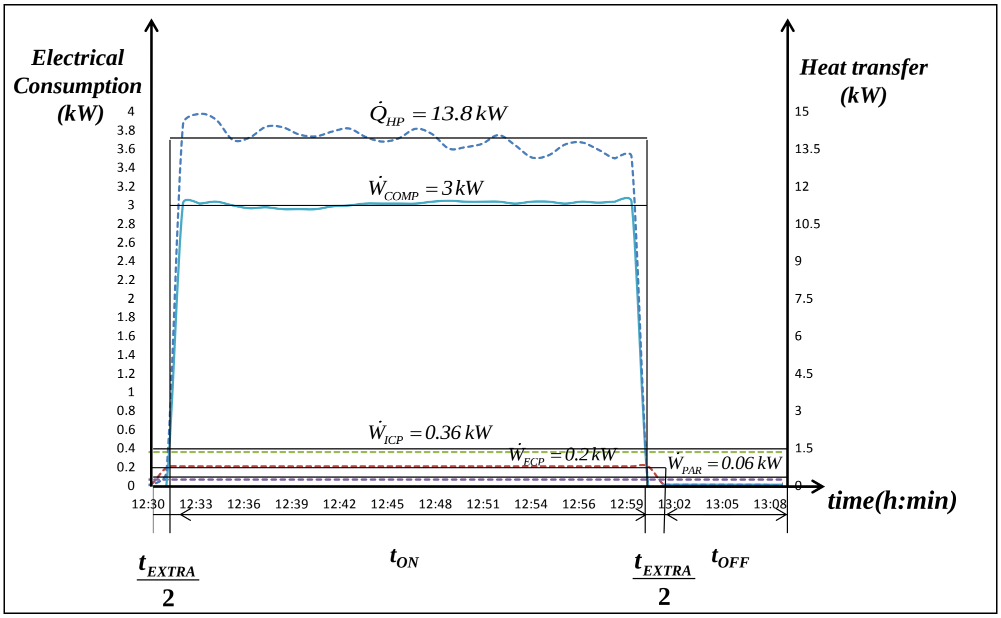


Figure 4 – Power consumption of the system components, for an ON/OFF cycle of the heat pump.

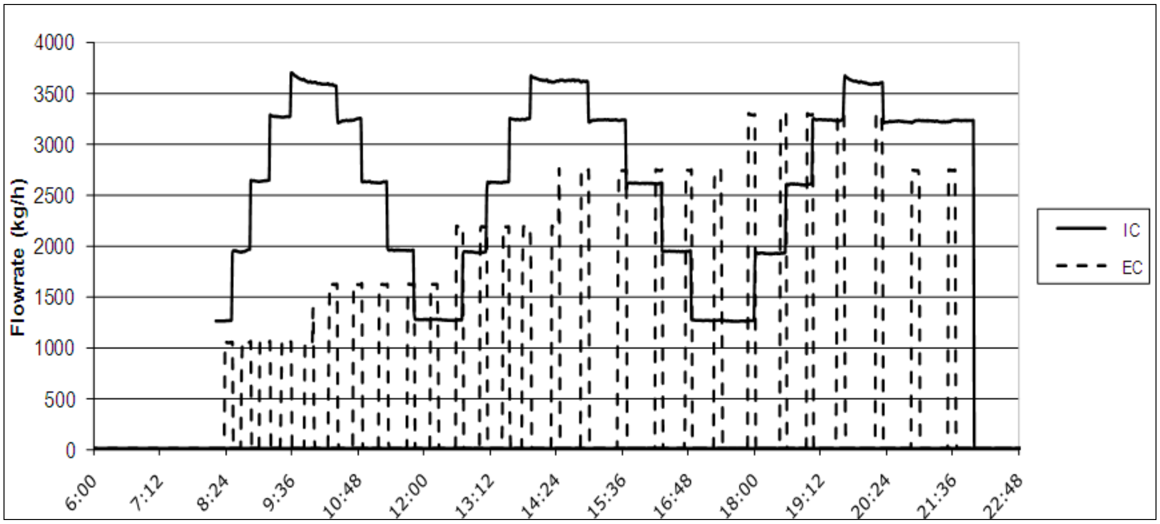
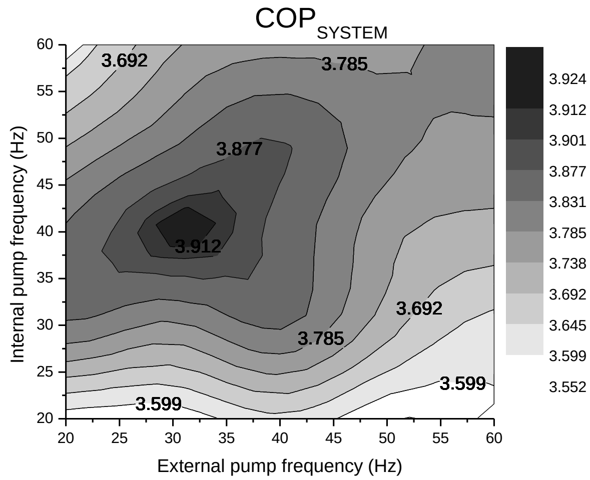
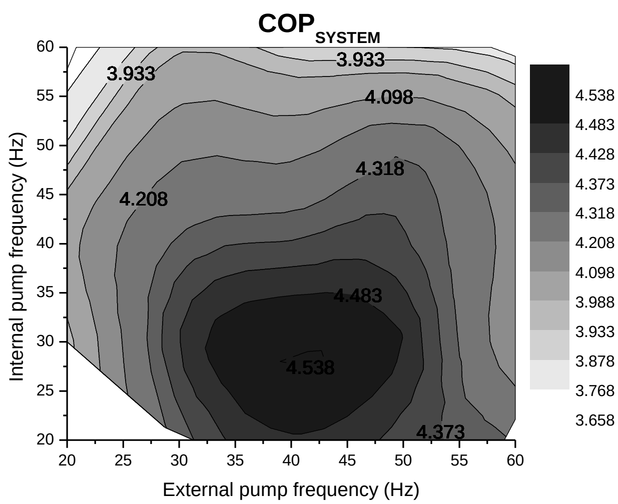


Figure 5 - Frequency variation strategy in heating mode.



a)



b)

Figure 6 - COP$_{SYSTEM}$ quasi-steady state maps as a function of pump frequency a) Heating mode b) Cooling mode.

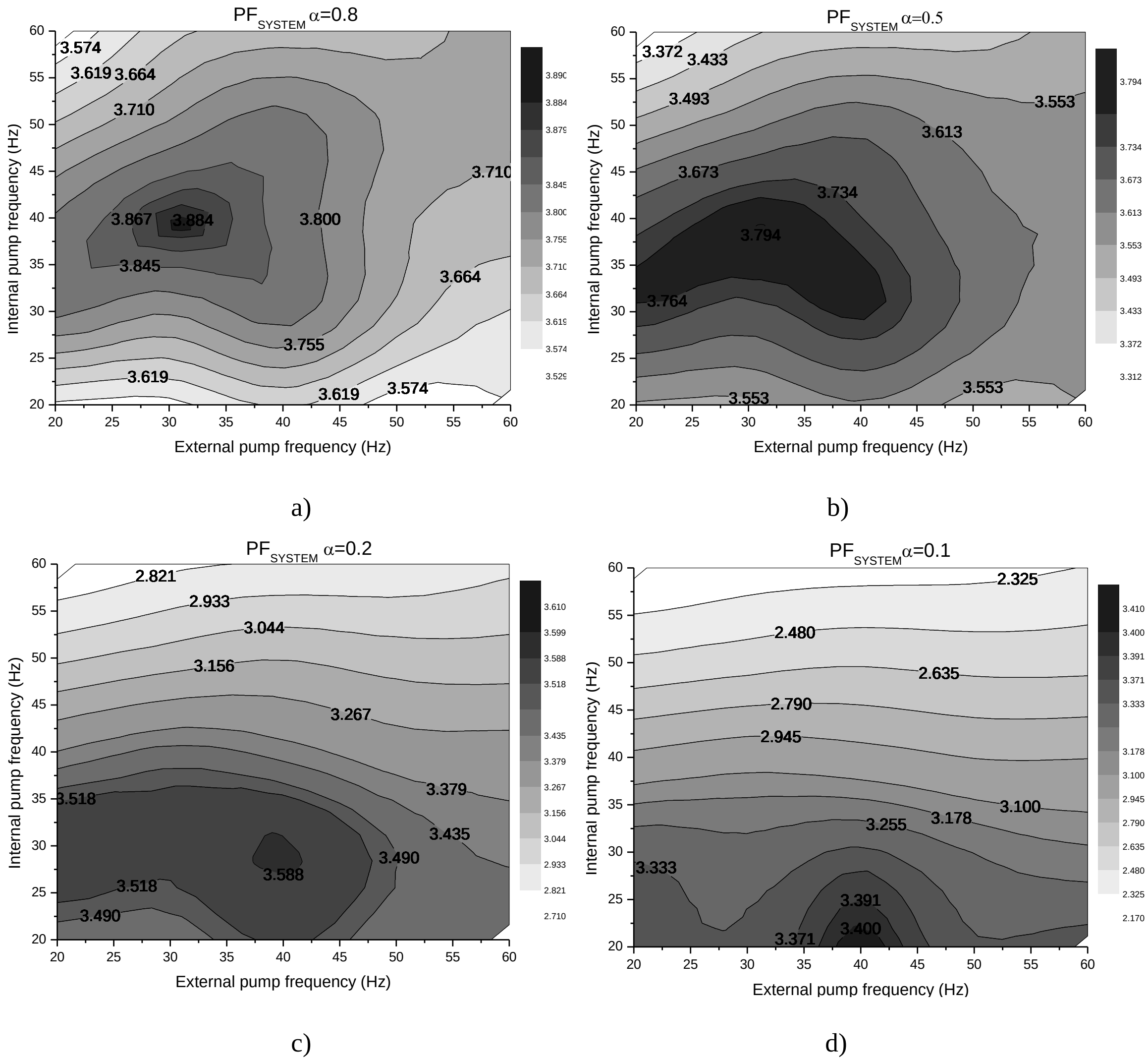


Figure 7- PF_{SYSTEM} maps as a function of pump frequency at heating mode: a) α =0.8, b) α =0.5,c) α =0.2, d) α =0.1.

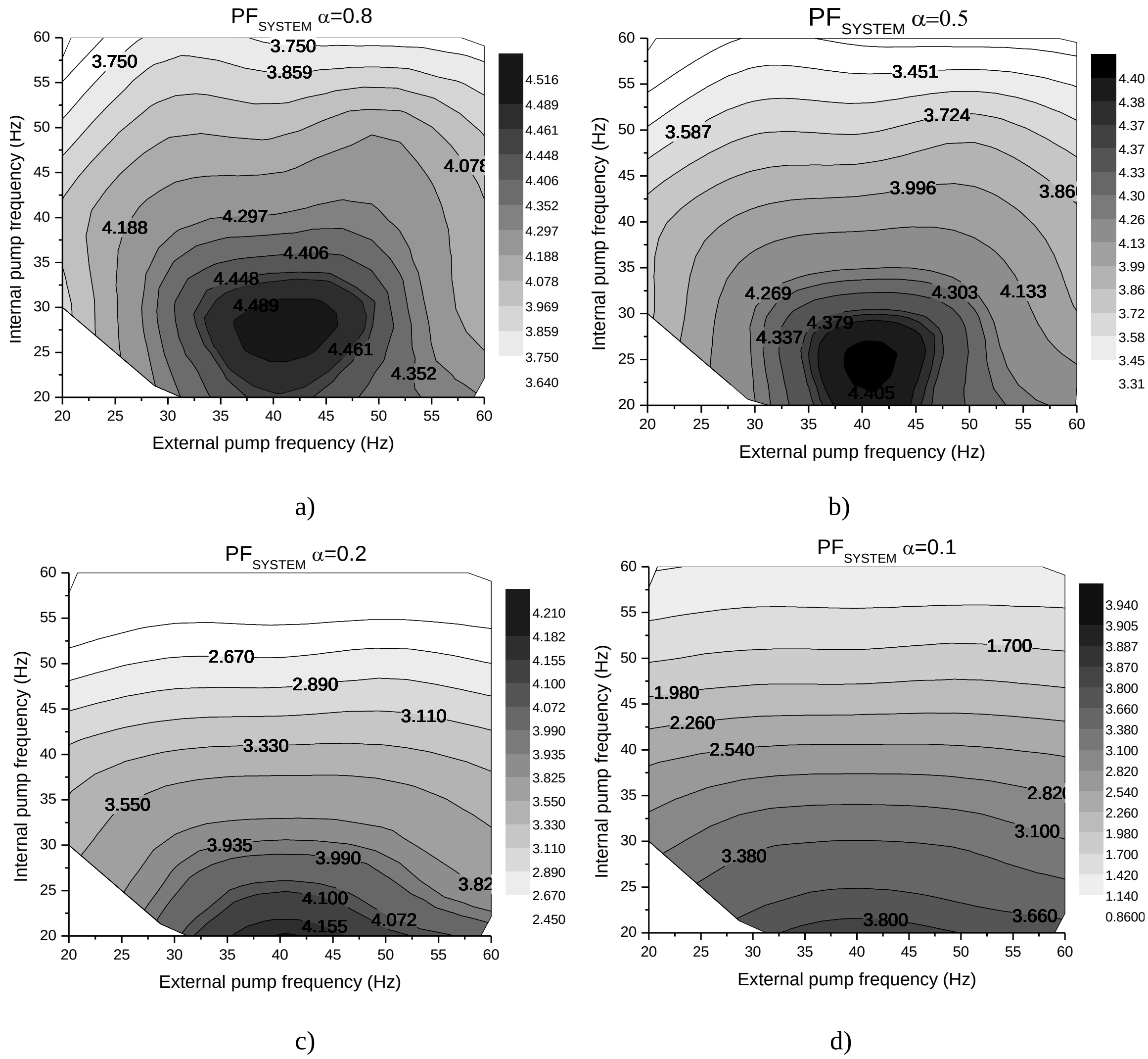


Figure 8 - PF_{SYSTEM} maps as a function of pump frequency at cooling mode: a) α =0.8, b) α =0.5,c) α =0.2,d) α =0.1 .

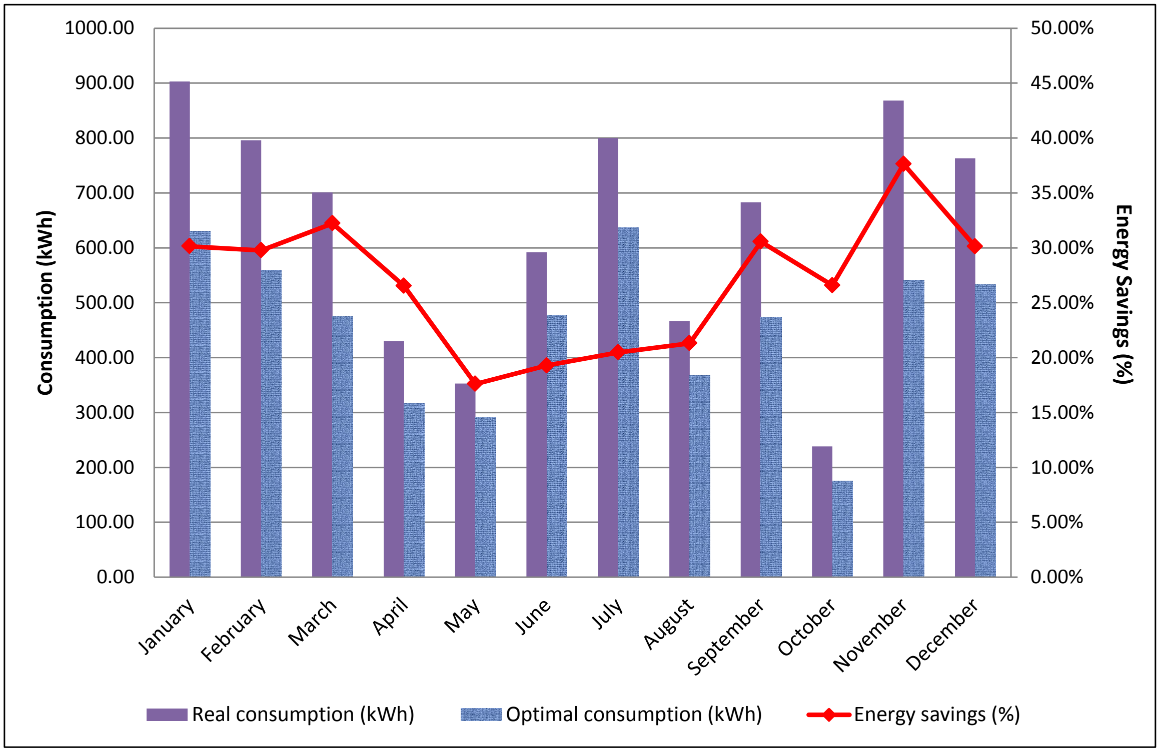


Figure 9 – Energy savings estimation for one year of operation.

***Highlights (for review)**

>Impact of the energy consumed by auxiliary equipment (circulation pumps) >Three steps optimization methodology carried out on site. >Optimal operating frequency as a function of instantaneous thermal load.>Obtained results indicate 28% energy savings potential.